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REPORT DOCUMENTATION PAGE

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1. REPORT NUMBER: W.P. No. 196-6

2. GOVT ACCESSION NO.

3. RECIPIENT'S CATALOG NUMBER

4. TITLE (and Subtitle): (6) On Quantizing Ride Comfort and Allowable Accelerations. (U)

5. TYPE OF REPORT & PERIOD COVERED

6. PERFORMING ORG. REPORT NUMBER

7. AUTHOR(s): (10) Peter R. Payne

8. CONTRACT OR GRANT NUMBER(s): (15) N00167-76-M-8393

9. PERFORMING ORGANIZATION NAME AND ADDRESS: Payne, Inc. Annapolis, Maryland 21401

10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS

11. CONTROLLING OFFICE NAME AND ADDRESS

LEVEL

12. REPORT DATE: (11) July 1976

13. NUMBER OF PAGES: 15

14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office): DTNSRDC Bethesda, Maryland 20084 (12) 18

15. SECURITY CLASS. (of this report): Unclassified

15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Unlimited and approved for Public release.

DTIC ELECTE MAR 3 1980 S D C

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

(14) WORKING PAPER-196-6

18. SUPPLEMENTARY NOTES

This report used by OP96V in their study: Advanced Naval Vehicle Concepts Evaluation.

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Advanced Naval Vehicle Concepts Evaluation
ANVCE
Technology Assessment
Ride Quality of Naval Vehicles
Motion
Velocity Changes

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

When the motion of a vehicle includes "shocks" or impulsive velocity changes, R.M.S. acceleration has no relation to crew comfort or injury. Existing (R.M.S. "g") methods of ride assessment can show lethal accelerations as being perfectly safe. They are also said to be invalid when the acceleration "crest factor" (peak/R.M.S.) exceeds 3, which is often the case for high-speed marine vehicles. This paper presents methods of evading these difficulties, using fairly well established biodynamic modelling techniques, →

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N00167-76-M-8390

Working Paper No. 196-6
July 1976

ON QUANTIZING RIDE COMFORT AND ALLOWABLE ACCELERATIONS

for

David W. Taylor Naval Ship Research & Development Center

Bethesda, Maryland 20084

Contract No. N00167-76-M-8390

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ON QUANTIZING RIDE COMFORT AND ALLOWABLE ACCELERATIONS

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Abstract

When the motion of a vehicle includes "shocks" or impulsive velocity changes, R.M.S. acceleration has no relation to crew comfort or injury. Existing (R.M.S. "g") methods of ride assessment can show lethal accelerations as being perfectly safe. They are also said to be invalid when the acceleration "crest factor" (peak/R.M.S.) exceeds 3, which is often the case for high-speed marine vehicles. This paper presents methods of evading these difficulties, using fairly well established biodynamic modelling techniques, and an extension of Allen's "shock tolerance" concept. Among other advantages, the method "automates" the assessment of ride quality, so that personal judgments are not involved, and the relative ride quality of different vehicles can be placed on a quantitative basis.

Summary

A key requirement for a meaningful evaluation of any new marine vehicle is the need for a clear, unambiguous definition of ride quality. By this we mean a meaningful measure of the "roughness" of the ride; not a measure of each crew member's performance capability under specified ride conditions. No such definition seems to exist in the literature. The best known set of "allowables" for random vibration, ISO-2361[1] specifically excludes accelerations which have a "crest factor" (CF = Peak/RMS) in excess of 3.0, whereas naval vehicles may experience crest factors in excess of 10. Moreover, ISO is concerned with "safe" accelerations for the population in general, rather than a presumably fit and well-motivated Navy crew. Finally, all authorities seem to agree that ISO-2361 cannot be used at all when the input acceleration contains a series of "impacts" or "impulsive velocity change" spikes. While such a "spike" may be injurious, or indeed lethal, it need not have much effect on the RMS acceleration. And it's the RMS acceleration which is considered in the ISO-2361 type of assessment.

This paper describes a method of assessing the ride quality of naval vehicles which is intended to avoid these limitations, and establishes two limits; one for random vibration having various degrees of severity, and one for impacts or "impulsive velocity changes."

Two separate indices are proposed in this paper to designate a vehicle's habitability for a given acceleration-time history. These indices are:

- Vibration Ride Quality Index (VRQI)
- Impact Ride Quality Index (IRQI)

The proposed limits for VRQI are as follows:

Limit	Description	VRQI	ISO-2361 (For Frequencies Greater Than 1 Hz) Nearest Corresponding "Exposure Limit"
(A)	Severe, less than 1 hour	0.5	1 hour
(B)	Tolerable; less than 1 hour	0.2	4 hours
(C)	Long-term, Severe	0.2	4 hours
(D)	Long-term, Tolerable	0.1	16 hours

For reference purposes, VRQI = 2.3 would roughly correspond to a one-minute endurance limit to sinusoidal vibration, as determined in the laboratory (Appendix I).

Impact Ride Quality (IRQI) must be less than unity to meet the proposed limits, and a time-domain calculation must be carried out to obtain a plot of peak force exceedances in a dynamic model against frequency of occurrence. This "exceedance plot" must fall below an "allowable limit" exceedance plot. The IRQI is a measure of severity, relative to this limit.

While it is normal to be conservative in specifying "allowables" for crews, conservatism can be extremely expensive. For example, reducing allowable acceleration by 30% might double the basic cost of a vehicle, because of the increase in size required.[27] With this in mind, the limits defined herein are deliberately severe by generally accepted standards. It is hoped that work will be funded to compare them with data from existing naval vehicles, with a view to modifying the limits if necessary. But it is anticipated that the basic dynamic models will not be changed by any such modification; only the limit RQI values.

Introduction

Despite the amount of laboratory research undertaken in the last three decades, the definition of human tolerance to vibration is still very tentative. This is for three main reasons:

- Mankind is very variable, and even an individual is not always consistent.
- The interaction between man's comfort or task performance and his acceleration environment is extremely complex.
- Relatively little effort has been devoted to the engineering problem of establishing rational tolerance limits for vehicles.

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The standard by which all other limits are compared is, of course, ISO 2631[1]. A careful reading of this standard, and the various commentaries upon it which have appeared, shows that there is still a long way to go before a reliable and consistent set of limits can be established for naval vehicles. It is the purpose of this present document to present limits which are at least consistent, so that they can be applied uniformly and unambiguously, even though the limits may not be entirely "correct" in an absolute sense.

To better understand the problem, it may be helpful to briefly review the history of an analogous, but more mature problem; that of human tolerance to short period (pulse-type) acceleration. For acceleration in the spinal direction, definition of accurate tolerance limits is of critical importance in the design of aircraft ejection seats. Thus a great deal of research has been performed. It is also a simpler problem than that of defining vibration tolerance, because the limit or "end point" - vertebral fracture - is known and fairly well understood.

Prior to about 1960, the definition of human tolerance to short period, linear acceleration was poorly organized. The various "tolerance curves" in use did not agree with each other and sometimes led to incorrect requirements being placed on the designer. For example, it was generally thought - quite erroneously - that "jerk" (the "rate of onset" of acceleration, d^3x/dt^3) was physiologically important, and this misconception caused a great deal of needless trouble and expense.*

Yet the problem of defining an acceptable tolerance curve was quite a simple one to solve, when looked at from the viewpoint of a dynamicist, rather than that of an M.D.[28,29] Von Gierke, Hess, Latham, Coerman, Kornhauser, and other workers appreciated that quite simple dynamic models of the human body could be adequate for interim engineering purposes, once the correct spring rates and damping coefficients could be discovered. Payne first analyzed the situation and proposed specific criteria; tentatively in 1961[2] and more definitively (using all the data then available) in the period 1962-63[3,4,5]. This model for upward acceleration of a seated man (Figure 1) is now the MIL Spec Standard[2] for the design of escape systems, and an Air Standardization Coordinating Committee (ASCC) Standard as well. While employed principally in the design of aircraft escape systems, it is also used in fields far removed, such as the design of snowmobile seats and suspensions.

Use of the model is very simple. Any acceleration-time history, no matter how complex, can be imposed upon it. The output is a single number, the "DRI", which is proportional to the peak load in the model's "spine" during that acceleration. From Figure 1, we then obtain the percentage of vertebral fractures which will be experienced with that particular DRI, and decide whether to modify the acceleration input. It is possible to intelligently trade-off the predicted incidence of vertebral injury against, for example, the predicted death rate from impacting an airplane structure during aircraft escape.

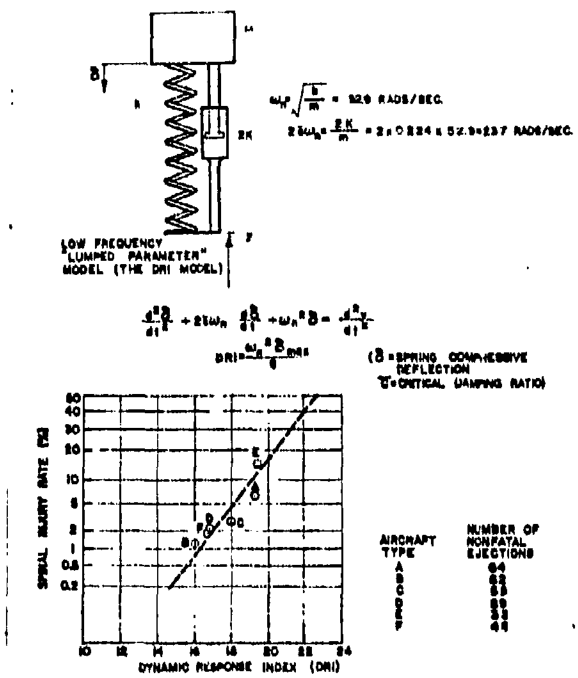


Figure 1. MIL-S-9478A(USAF)[2] and ASCC Standard seated man model for short period upward acceleration. The spinal injury correlation is from Reference 34.

One of the model's principal advantages is often overlooked. There is no room for "judgment", or argument; the "rules" cannot be changed to produce a more favorable DRI. Any engineer using the model as an evaluator will come to precisely the same conclusion as any other engineer. All comparisons are objective, in happy contrast to the extended (and necessarily unresolvable) debates which took place before the model's introduction.

The same type of approach is recommended in the ISO-2631**, when it is desired to "characterize the vibration environment with respect to its effects on man by a single quantity. . ." But the recommended weighting network is not well adapted to the types of vibration and shock which are experienced in naval vehicles. Briefly, the difficulties are as follows:

(a) As indicated in Figure 2, high-speed ship acceleration-time histories can be very "spikey", containing frequencies much higher than the 80 Hz upper limit of ISO-2631. "Crest factors" (peak/RMS) are typically higher than the maximum values of 3.0 for which ISO-2631 is considered valid.† For

* The experimenters who "identified" it were really seeing "rise-time" effects.

** Reference 1, page 5, second column, second paragraph.

† Reference 1, page 3; end of Section 3.3.

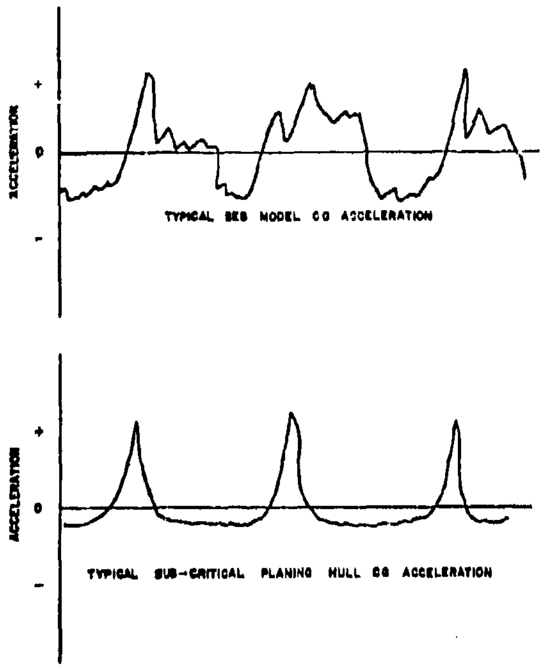


Figure 2. Acceleration-time histories for typical high speed naval vehicles.

example, some typical SES model tests give CF = 8 - 10. Some open water planing boat measurements give CF = 6.

(b) Many vehicles experience slamming, in which the acceleration-time history approaches an impulsive velocity change which cannot be analyzed spectrally.[7] And to average a severe slam over a period of time is obviously meaningless.

(c) An amplitude spectrum without phase information is mathematically meaningless.

A considerable number of investigators are considering these problems at the present time. See, for example, References 8 - 10. The general concensus seems to be that a lumped parameter dynamic model (of the type shown in Figure 1) is a realistic way of evaluating vibration environments with high "crest factors" and impulsive velocity change "shocks." Figure 3 is an example of one such approach, due to Allan.[16]

In the next section, we examine these problems in more detail, as they reflect on the engineering assessment of a marine vehicle's "ride."

The Problem of Impulsive Accelerations

The ISO-2361 tolerance curves are based primarily on experimental investigations with sinusoidal vibration. Their application to nonsinusoidal vibration is rather tentative, and in some ways, quite arbitrary. To the credit of the original drafters, however, the limits seem to "make sense" as they are more and more compared with data from operational situations, provided the crest factor is not too high, and that impulsive accelerations do not occur. This obviously implies some form of regularity in the missing phase angle spectrum.

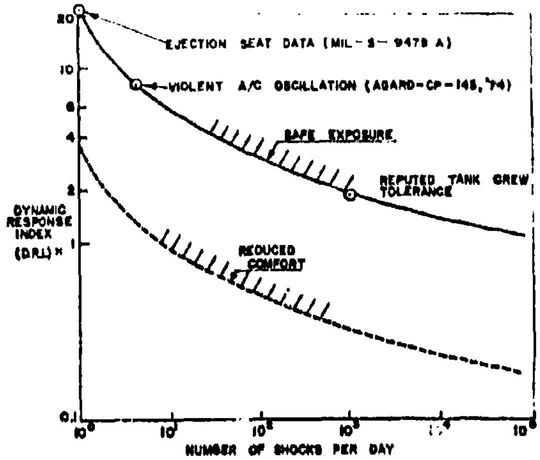


Figure 3. Possible approach for acceptable levels of repeated shocks [Allen[16]].

*From Figure 1.

Most high-speed ships experience occasional - sometimes frequent - "slamming" or "pounding" events which are quite unlike the "normal" vibration considered in ISO-2361. One way to examine this problem is to consider the case of any acceleration-time history which is composed solely of such impulsive accelerations. Such accelerations could be experienced by, for example, a subcritical planing hull pounding from wave flank to wave flank, and otherwise being out of the water. This is a feasible mode of operation, particularly in off-shore racing.

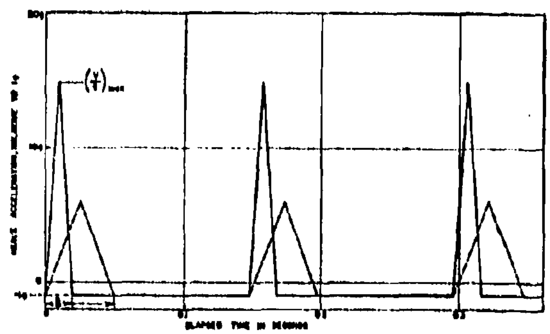


Figure 4. Idealized acceleration-time history for 40 knots in 10-foot wave lengths. $2\Delta v = 4.76$ ft/sec. (The spike base time δ will depend upon the details of the impact and the frequency response of the instrumentation and its structural attachment.)

An idealized acceleration time history for such motion is given in Figure 4.[7] The actual values of $\ddot{y}_{MAX}$ and δ which would be recorded in such a situation would depend upon:

- The frequency response of the accelerometer.
- The "hardness" of the structure to which it was attached.
- The details of the various structural resonances.
- The hydrodynamics of the impact.

Only the last of these is "real", in the sense that it influences a human occupant's perception of the acceleration. This is because, for very short duration accelerations, man is sensitive to velocity change ($= \frac{1}{2}\ddot{y}_{MAX}\,\delta$) rather than the magnitude of $\ddot{y}_{MAX}$[3-6]. Typical trajectory parameters, which are independent of wave height, are given in Figure 5.

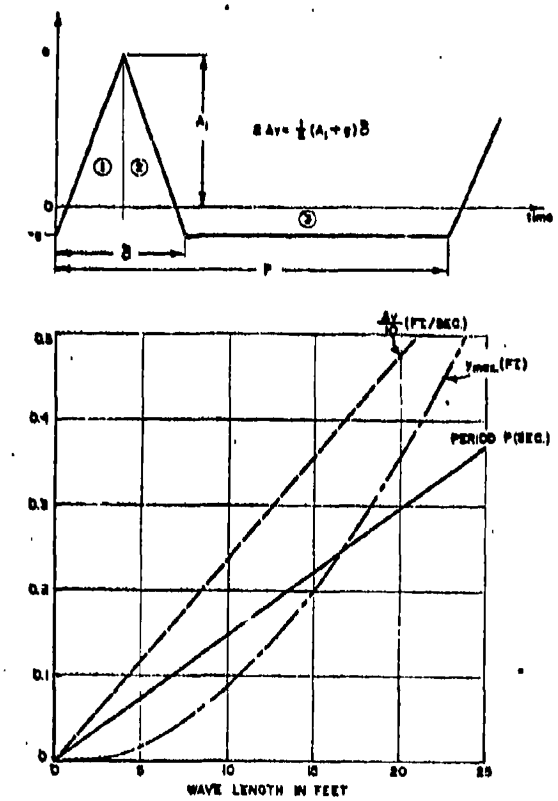


Figure 5. Idealized trajectory parameters for a boat speed of 40 knots. [Δv is the vertical (downwards) velocity of the boat just prior to impact, so that the total vertical velocity change is 2Δv. y_{MAX} is its maximum trajectory height between impacts, measured above the impact plane.]

Now, how do we assess the tolerability of Figure 4? As a first guess, we might decide that, since it is periodic, we can use the ISO-2361 criteria. We therefore determine its RMS value and its crest factor. It's easy to show[7] that

$$\left(\frac{\ddot{y}}{g}\right)_{RMS} = \sqrt{\frac{2}{3}\left(\frac{A_1}{g}\right) - \frac{1}{3}} = \sqrt{\frac{4}{3}\frac{P}{\delta} - 1} = \sqrt{\frac{8}{3}\frac{\Delta v}{g\delta} - 1} \quad (1)$$

This is a very unsatisfactory result, because the RMS depends upon the width (δ) of the acceleration spike. We know that 2Δv = 1 ft/sec is perfectly safe. But our RMS criterion implies that it's not, if δ is very small, i.e., if

δ =	.01	.001	.0001 seconds
$\ddot{y}_{RMS}$ =	1.77g	6.36g	20.33g

This is obviously not so, because a human subject could not perceive any difference among the three pulses. And the error is not due to high values of the "crest factor", incidentally, which for this time history is given by[7]

$$CF = \frac{A_1}{\ddot{y}_{RMS}} = \frac{1}{\sqrt{\frac{2}{3} - \frac{1}{3}\frac{g}{A_1}}} = \sqrt{\frac{4\Delta v - g\delta}{\frac{8}{3}\Delta v - g\delta}} \quad (2)$$

For A_1 =	2g	10g	∞
CF =	1.41	1.26	$\sqrt{3/2}$ = 1.23

These are very low values; much less than the upper bound of CF = 3.0 specified in ISO-2361. We therefore have to conclude that the ISO approach is not valid for this kind of periodic acceleration.*

A second possible approach is to assume that each impact event occurs by itself, after the effect of the previous one has damped out. We can then evaluate it by determining its spectral content, and following the ISO-2361 methodology.

The spectrum of a triangular pulse a(t) is given by its Fourier transform

$$F(i\omega) = \int_{-\infty}^{\infty} e^{-i\omega t}\, a(t)dt = 2\Delta v\left[\frac{\sin\frac{\omega\delta}{4}}{\frac{\omega\delta}{4}}\right]^2 \quad (3)$$

This is plotted nondimensionally in Figure 6. Again we see that the "real" spectrum, as a function of frequency, depends upon the value of the pulse width time δ. So when we evaluate it in accordance with ISO-2361, we get totally different results for different values of δ, as shown in Figure 7.

If we now turn to the DRI method[2] of evaluation, we find that[18]

$$DRI = \frac{\omega(2\Delta v)e^{-\tau\phi/\eta}}{g} \quad (4)$$

where

$$\eta = \sqrt{1 - \tau^2} \qquad \phi = \sin^{-1}\eta$$

For the DRI model (Figure 1)

$$DRI = 1.207(2\Delta v)$$
$$= 24.1 \text{ for } 2\Delta v = 20 \text{ ft/sec}$$

From Figure 1, this corresponds to a vertebral fracture rate in excess of 50%. So we have a situation where an acceleration which will cause grave bodily injury is "tolerable for one minute" under the ISO criteria. Clearly an unacceptable situation.

*We also have to conclude that the "Crest Factor" is an arbitrary parameter which has no mathematical meaning.

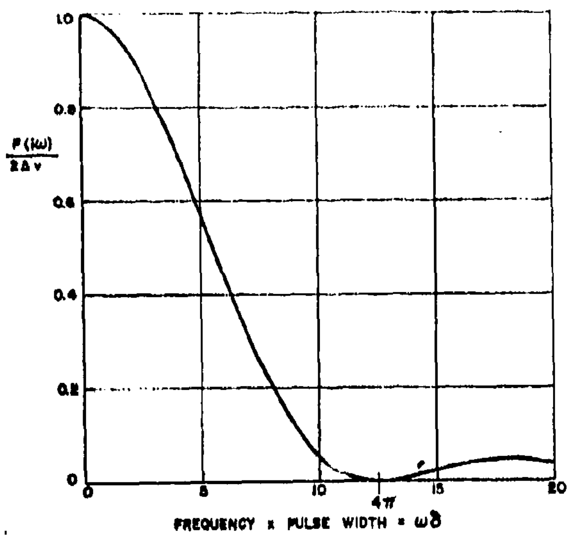


Figure 6. Partial spectrum of a triangular pulse (zeros occur at $\omega\delta = 4n\pi$, n integer).

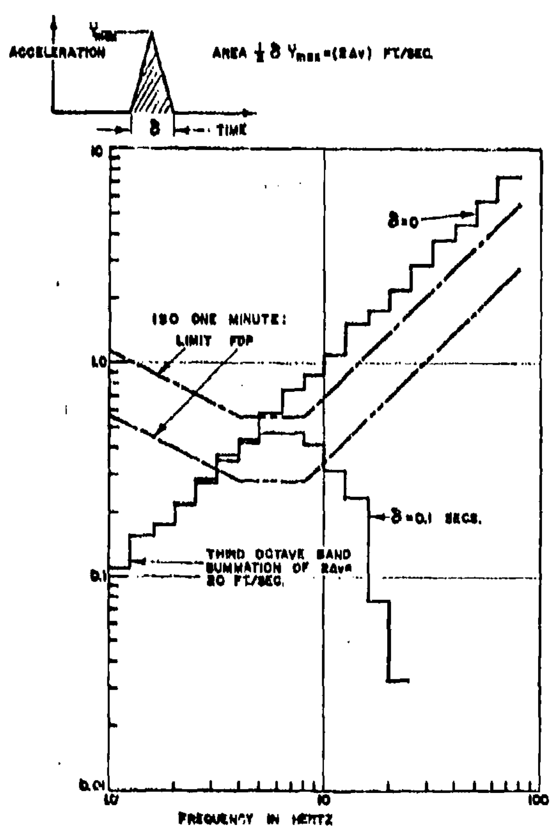


Figure 7. ISO-2361 evaluation of a 20 ft/sec velocity change impulse for two different values of the duration (δ).

To add emphasis to this point, let us suppose that we had a spectrum which was uniform and infinitely wide.* This could represent one of two inputs $\ddot{y}_c(t)$

(1) A Dirac impulse which could kill a man, or

(2) White noise which would be imperceptable to him.

Obviously, then, the spectrum by itself is meaningless without phase information.

Our analysis has been confined to mathematically simple acceleration-time histories which permit mathematical rigor, and we have seen that the ISO criteria cannot be applied to them in any meaningful way. If we now consider an acceleration of the type shown in the upper half of Figure 2, we find that the same difficulties arise, but are more difficult to spot, because of the greater complexity of the input. This is the trap into which we have fallen! Because a typical ship vibration is too complex to analyze rigorously, we have "guessed" at a methodology, by loose analogy with sinusoidal vibration, without attempting to validate it theoretically. And validation or rejection by experiment, when the "instrumentation" is as imprecise as the subjective reactions of large numbers of people, is likely to be time consuming at least; if not impossible. Note that the originators of the ISO Standard were well aware of this problem, and emphasized that the Standard is not applicable when the vibration deviates significantly from sinusoidal.

It's therefore imperative to use a more meaningful method of ride evaluation. Fortunately, the linear dynamic model assessment method is available, and is already well proven for just those aspects which cause so much difficulty with the RMS approach. In addition to solving these difficulties, dynamic models can at least reproduce all the other features of the ISO assessment method, and are probably superior, in that they account (albeit imperfectly) for phase effects.

In the following sections, we describe and define the appropriate dynamic models. The relevant sinusoidal excitation theory may be found in Appendix II, and the transient theory in References 3-6 and 18.

The Proposed Ride Quality for Frequencies Above One Hertz

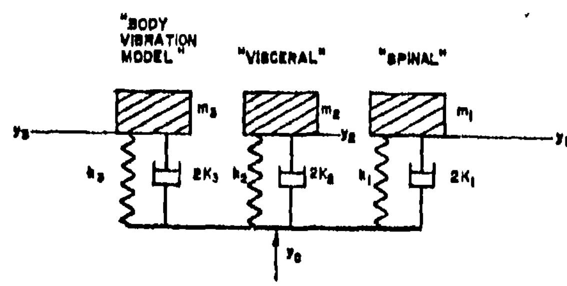


Figure 8. The proposed model for frequencies above one Hertz.

*This example was suggested by Dr. Charles W. McCutchen in a personal communication.

The proposed model has three degrees of freedom in parallel as indicated in Figure 8. System (1) is the MIL-S-9478A model of Figure 1 and may be loosely referred to as a "spinal" model. System (2) has a lower frequency, and is intended to account for the various "visceral modes." The last, high frequency mode represents "body vibrations." It is not identified with any particular physiological system or characteristic, and its parameters are selected solely to agree with existing "vibration tolerance" boundaries. A given ISO curve roughly corresponds to one of the mass accelerations (RMS) $\ddot{y}_1$, $\ddot{y}_2$, $\ddot{y}_3$ being equal to the appropriate critical value, e.g.

$$\ddot{y}_1 = \ddot{y}_{n_{crit}} \text{ and } \ddot{y}_2 \le \ddot{y}_{n_{crit}},\ \ddot{y}_3 \le \ddot{y}_{n_{crit}} \quad \text{(RMS)}$$

The model coefficients are as follows:

$\omega_1 = 52.9$ rads/sec $\zeta_1 = 0.224$ (The "spinal" or DRI model)

$\omega_2 = 25.1$ rads/sec $\zeta_2 = 0.4$ (The "visceral" model)

$\omega_3 = 52.9$ rads/sec $\zeta_3 = 1.0$ (The "body vibration" model)

Critical Mass Accelerations (RMS)

(Limit A)	"Severe, less than one hour"	$\ddot{y}_{n_{crit}} = 0.5g$
(Limit B)	"Tolerable, less than one hour"	$\ddot{y}_{n_{crit}} = 0.2g$
(Limit C)	"Long-term severe"	$\ddot{y}_{n_{crit}} = 0.2g$
(Limit D)	"Long-term tolerable"	$\ddot{y}_{n_{crit}} = 0.1g$

Apart from the change at one hour, the ISO-2631 notion of time dependence is not employed in this model. The concept of performance degradation with time has not been supported by laboratory experiments (see, for example, Von Gierke[11,19] and Maslen[12]) and there does not seem to be any reason to suppose that trends postulated for the general population apply to well-motivated Navy personnel at sea.

Comparison of the Proposed Vibration Limits With ISO-2631

A typical limit generated by this model, the limit for sinusoidal vibration, is shown in Figure 9. It can be seen that the visceral model controls up to about 4.6 Hertz, then the spinal model controls up to 11.7 Hertz. Above about 30 Hertz, the highest frequency body vibration model has the same slope as the ISO limits.

A particular advantage of the model is that one parameter $\ddot{y}_{n_{crit}}$ governs the "severity" of the limit, all other parameters being fixed. The tentative limits are compared with the ISO exposure limits in Figure 10.

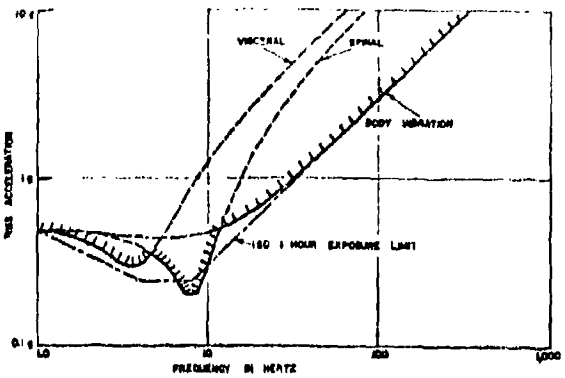


Figure 9. Model response to sinusoidal vibration - "severe, but tolerable; less than one hour."

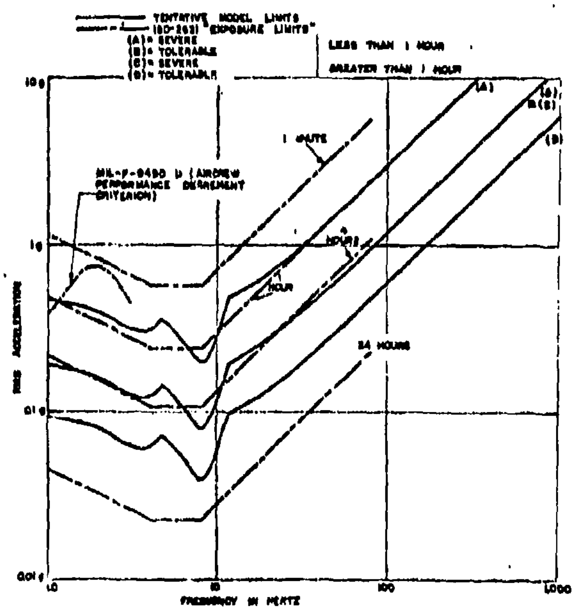


Figure 10. The tentative limits compared with ISO-2631 exposure limits of sinusoidal vibration.

Quantizing "Vibration Ride Quality"

Each RMS mass acceleration must be less than the critical value $\ddot{y}_{n_{crit}}$ specified above. For each degree of freedom in the model, there will be a "vibration ride ratio"

$$\text{(VRR)} = \frac{\ddot{y}_n}{g} \quad \text{(RMS)}$$

The "vibration ride quality index" (VRQI) is the highest value of (VRR) among the three systems.

A value VRQI = 1.0 would be very severe; twice as much as the proposed "Severe, less than one hour" limit. A value of 2.3 would correspond to the "one minute endurance limit" established by Ziegenruecker and Magid[17]. The proposed limits are:

		VRQI
(Limit A)	"Severe, less than one hour"	0.5
(Limit B)	"Tolerable, less than one hour"	0.2
(Limit C)	"Long-term severe"	0.2
(Limit D)	"Long-term, tolerable"	0.1

Shock or Impact Criteria

The number and magnitude of shocks which can be tolerated in unit time is defined by an extension of Allen's[16] approach, as shown in Figure 11. This method utilizes the DRI or "spinal" model (system 1 in Figure 8) to count the number of times the DRI exceeds various thresholds.

$$DRI = \frac{\omega_1^2 \delta_{1_{MAX}}}{g} = 86.961\ \delta_{1_{MAX}} \quad (\delta \text{ in feet})$$

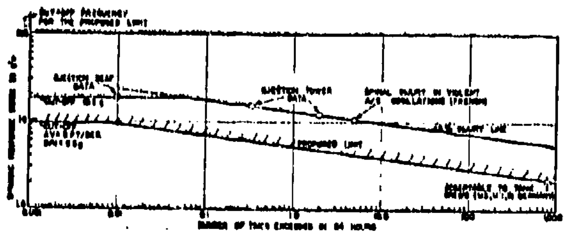


Figure 11. Tentative limits for the Dynamic Response Index.

Each maximum must be counted, i.e.

Each value of δ_1 when $\dot{\delta}_1 = 0$, $\ddot{\delta}_1 < 0$.

An exceedance plot is then readily produced. To be tolerable, it must everywhere be below the limit in Figure 11. The equation for the limit is

$$(DRI)_{LIM} = 9.5 \qquad (0.001 < N < 0.00776)$$

$$(DRI)_{LIM} = \frac{5.173}{N^{.1251}} \qquad (N < 0.00776)$$

Where N = number of exceedances per 24 hours. For N < .001, no limit is proposed, implying that the critical DRI may be exceeded once every thousand days of continuous operations, or about once every ten years of elapsed time. This is to avoid logical conflicts arising with conventional statistical analysis of ride data.

For impacts which occur 1 to 10 times a second (N = 86,400 to 864,000), this limit can permit a somewhat more severe $\ddot{y}_c$ RMS value than the most severe vibration limit for one hour or less. It implies 0.87g RMS at one Hertz and 0.45g RMS at 10 Hertz. From a research point of view, this agreement is remarkably good, considering how the repeated shock limit was established, and gives a considerable increase of confidence in the approach. From a "specification" point of view, we can avoid any difficulty by simply requiring the lowest of the two limits to be used, when they conflict.

More research clearly needs to be done in this area.[26] How do we "join" the two different criteria in a logical manner? Should we have different allowable DRI levels for less than and more than one hour? Perhaps time dependency should be even more detailed. Perhaps we can evade the need for the DRI criterion by imposing additional constraints on the vibration model output. At a deeper level, what is the fatigue mechanism?

Characterizing "Impact Ride Quality Index" (IRQI)

We define the "Impact Ride Quality Index" as

$$IRQI = \left.\frac{B(N)}{L(N)}\right|_{MAX}$$

$$= \text{maximum value of } \frac{\text{DRI at a particular frequency}}{\text{The limit at that frequency}}$$

As the examples of Figure 16 show, this is a very simple concept* in practice; in fact, simpler to use than to define.

The Proposed Low-Frequency Model

Even though a ship may be travelling slowly, and there are no high-frequency acceleration components, it is a matter of common experience that its occupants can experience a loss of efficiency, due to two effects:

(a) Ship motion may make ordinary tasks more difficult. For example, walking, performing maintenance work, carrying loads. This is usually referred to as a degradation in motor performance.

(b) Ship motion may cause motion sickness, popularly known as "seasickness", and medically as "kinetosis."

These two phenomena are not necessarily separate and distinct. Motion sickness may cause clumsiness and poor coordination; exacting, close-up work may cause motion sickness in an environment which would otherwise be acceptable. Analogously, loss of an horizon reference can result in sickness.** Both phenomena are known to be associated with balancing mechanisms in the inner ear [Berry[20]].

It is also a matter of common experience that sailors adapt to their environment and that generally speaking, efficiency can return to near normal after a few days at sea unless motions are very severe. The "rolling gait" of the small boat sailor just ashore is a well known symptom of man's ability to adapt to walking on a rolling and heaving deck. In a small boat commencing a voyage in, say,

*An approach which was independently suggested by both K.R. Maslen[26] and E.G.U. Band of Payne, Inc.

**When automobiles were not an everyday experience from birth, one could often find people who could drive all day without any concern, but be quite uneasy as a passenger; and quite sick if they tried to read while being driven.

North Atlantic waters, it's typical for the crew to be sick for the first day or so, and from then on to be impervious to kinetosis in any reasonable sea state. On larger ships, the problem is more complex because the seas may be light for, say, the first week, and sickness may not develop until the first patch of rough weather causes the ship to move around.

People are also extremely variable. Some never get sick. Others are sick after months at sea, when there is a substantive change in ship motion (e.g., Admiral Lord Nelson). Even a given individual has a varying tolerance, depending on his morale, the nature of his last meal, and his environment of temperature, humidity, odor, horizon reference, and whether or not he is fatigued.

There does not appear to be any satisfactory quantized data on either motor performance or motion sickness incidence (MSI) in the Navy environment. O'Hanlon and McCauley[21] have produced kinetosis data for 600 subjects in the O.N.R. simulator, but this data is for unadapted young men in a laboratory environment. As a matter of interest, some of the Reference 21 data is replotted in Figure 12 to give a physical feel for the amplitudes (half peak to trough) involved, and to emphasize the importance of period. The published work of O'Hanlon and McCauley tells us nothing about motor performance degradation.

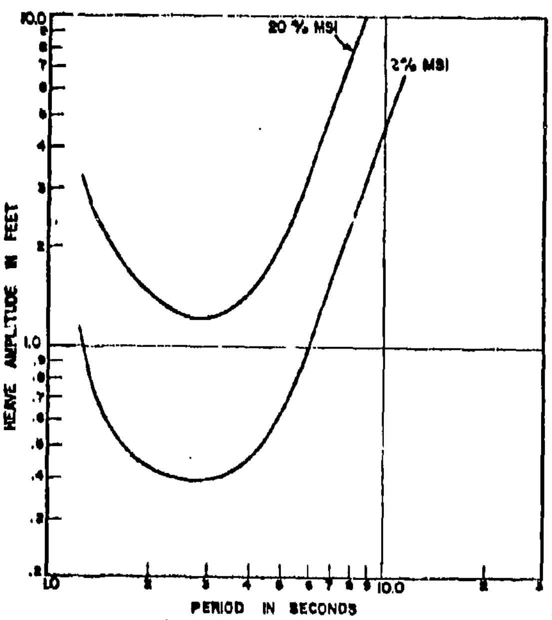


Figure 12. Curves of constant motion sickness incidence (MSI) for unadapted subjects within two hours after initial exposure to motion. (Adapted from Reference 21).

$$\text{MSI} = \text{Motion Sickness Incidence} = \frac{\text{Number of Subjects Sick}}{\text{Total Number Tested}}$$

Warhurst and Cerasani[22] [whose work was used by Hadler and Sarchin[23]] purport to give data on motor performance, but their results are inconclusive for the following reasons:

1. They assume that motion sickness is not involved. But the ship was on a two-week cruise ("the calmest trip we've ever had", according to one crew member) and experienced rough weather for only one four-hour period. It's possible that the motor performance degradation observed during this four-hour period was in part due to mild kinetosis, and that efficiency would have improved after a day or so.

2. They assume that only roll is important, ignoring pitch and heave. Anyone who has tried to work "up forward" in head or following seas will recognize the inadequacy of this assumption. Laboratory tests[24] seem to show that roll itself may not be important at all.

3. They purport to show that efficiency may improve with moderate roll. But their data does not support this; only the "motivation" factor shows this improvement. Did these moderate roll rates perhaps occur while returning to port?!

4. As they say themselves, their data are not statistically significant.

It seems clear that Warhurst and Cerasani were working with a very limited budget, and were hoping to get a "first rough cut" at the problem. Other workers have perhaps stretched the results more than the original authors would have desired. And their overall approach was certainly sound: to find out how sailors perform at sea, one should go to sea and see.

Generally speaking, roll motion is much more "visible" than heave or pitch, because of the horizon reference. So it's usual to empirically relate discomfort to it, as did Warhurst and Cerasani. Yet the limited amount of laboratory data available indicates that angular roll has little or no effect on kinetosis, and may not be important to motor performance. Figure 13, which illustrates angular effects on kinetosis, is taken from McCauley, et al[24] and seems to show that angular effects are of second order importance. McCauley, et al[24] also cite earlier authors[31,32] who have asserted that pitch and roll are relatively unimportant compared with heave motion because the angular acceleration aboard ships is generally very low.

It would therefore seem that local heave is the most important parameter, so that if

z = height of the ship's CG above an inertial reference plane

τ = ship pitch angle

θ = ship roll angle

x,y = longitudinal and lateral distances from the CG

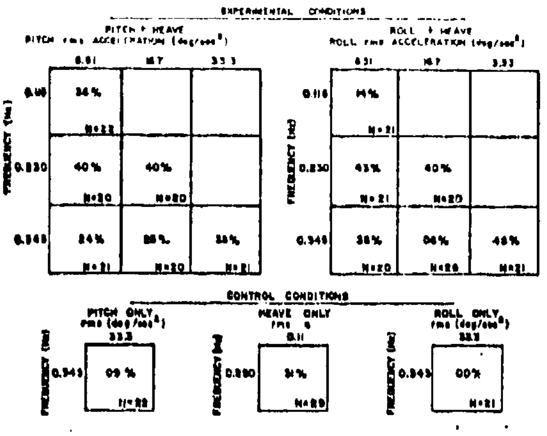


Figure 13. Motion Sickness Incidence (MSI) as a function of frequency and acceleration for pitch + heave, roll + heave, and for heave, pitch, and roll alone. All heave motions were at a frequency of 0.25 Hz and an RMS acceleration of 0.11g. Data from McCauley, et al.[24]

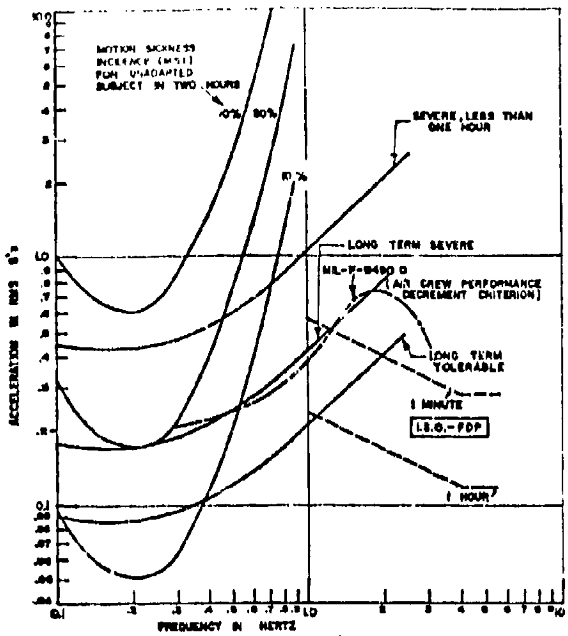


Figure 14. Various indicators of tolerable vertical acceleration.

then the motion of importance to a crew member located at x,y is:

$$h = z + x\tau + y\beta$$

Expressed as a local heave acceleration:

$$\ddot{h} = \ddot{z} + x\ddot{\tau} + y\ddot{\beta}$$

Thanks to the work of O'Hanlon and McCauley[21] we can relate h (or $\ddot{h}$) to the probability of sickness (MSI) for unadapted adult males. A summary of more recent work is given by McCauley, et al[24] including equations which permit MSI to be computed when acceleration amplitude, frequency and duration are known.

Of course, we cannot easily relate this data, obtained with inexperienced subjects, to experienced sailors, or to the motor performance of experienced or naive crew members. But it does give a lower boundary.

Another source of information is the "allowable" short-term acceleration curve defined in MIL-F-9490D for an "acceptable" degradation in the tracking performance of experienced aviators. Brumaghim's[33] presentation of this is compared with the MSI curves in Figure 14 and is seen to coincide with the 50% MSI (unadapted) data in the frequency range 0.1-0.3 Hertz. This seems reasonable.

The allowable acceleration defined in MIL-F-9490D is assumed to decrease as the duration increases, in a manner very similar to that of the ISO allowables. But Brumaghim[33] cites a number of experimental investigations which seem to contradict this, in that no performance degradation is found for test durations of up to six hours.

Based principally on Brumaghim's discussion, therefore, we propose, very tentatively, the following model for "long-term, severe":

$$\omega_L = \frac{\pi}{2}\ \text{rads/sec} \qquad \tau_L = 1.0 \qquad \ddot{y}_{L_{CRIT}} = 0.2g$$

The curve associated with this is shown in Figure 14 as "long-term severe." It's very odd that $\ddot{y}_{L_{CRIT}}$ = 0.2g the same value as "long term severe" for the higher frequencies. Does this mean, perhaps, that $\ddot{y}_{L_{CRIT}}$ = 0.5g would be a reasonable "severe, less than one hour" limit, and 0.1g the same for "long-term, tolerable?" When plotted in Figure 14, these limits don't look at all unreasonable for experienced sailors, so we might as well let them stand, lacking better data, until someone comes along with better figures. We then have the intellectually pleasing (but physically meaningless?) result that discomfort depends only on model acceleration and is independent of frequency, even below one Hertz. This has already been suggested by several workers, notably Jex, for the range above one Hertz.

Some physical feel for these limits, in the context of advanced marine vehicles, is given by Figure 15. This is an idealized calculation for a vessel which is small in relation to the wave length and is able to contour the surface by reacting negative acceleration loads as effectively as positive ones. Surprisingly, we see that the higher swells, because of their longer length, are more tolerable at a given speed than the smaller ones. The adverse effect on comfort of increasing ship speed beyond conventional values is clearly seen in Figure 15.

Future Work

To improve our knowledge of the relationship between ship motion and crew efficiency, it would be quite simple and inexpensive to go to the sea in

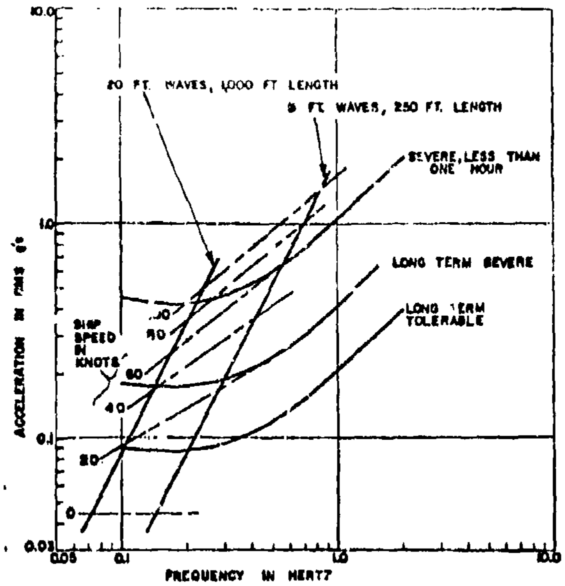


Figure 15. Idealized acceleration in regular sinusoidal head swells, compared with the proposed low-frequency limits.

various sized Navy ships and small craft and get direct readings in the real environment. The requirements would seem to be as follows:

1. A portable six-degree-of-freedom motion recorder. The existing PODAS would do, although real time processing of the data (average roll, pitch and heave amplitudes in the last five minutes, for example) would be highly desirable as well.

2. A convenient presentation of the space ordinates x, y, and z for all manned locations in the ship.

3. Standard questionnaires to be filled out by the crew when requested over the ship's address system.

4. Two or three simple physiological tests; two or three simple manual dexterity tests; a portable tracking task, and one or more physiologists to select and administer all of these.

5. An engineer to pull together all this data and analyze it on a real time basis (rather than after each voyage is over) so that questionable results can be immediately identified and re-analyzed as necessary.

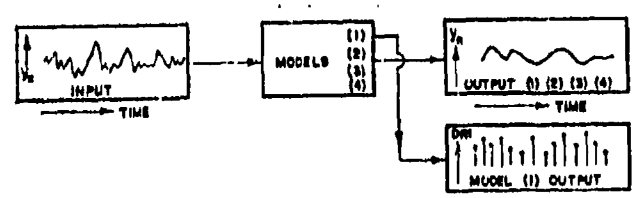
Conclusions

We have proposed a method of defining physiological "ride quality" limits. In decreasing order of severity, these limits are:

Limit	Physiological Description (Experienced Navy Crew)
(A)	Severe, less than one hour
(B)	Tolerable, less than one hour
(C)	Long-term, severe
(D)	Long-term, tolerable

One would normally design to meet limits (B) and (D), accepting the more severe conditions (A) and (C) for only a small percentage of the total operational profile, or if very substantial advantages (such as greatly reduced cost, perhaps) accrue from operation at the more severe limit.

The physiological effect of the vehicle's acceleration time history ($\ddot{y}_c$) for a given set of operating conditions is assessed by exciting (or "driving") four dynamic models with it, and observing the model output ($\ddot{y}_n$).



The basic model equation is as follows:

$$\ddot{\delta} + 2\zeta\omega_n\dot{\delta} + \omega_n^2\delta = \ddot{y}_c$$

$$\ddot{y}_n = \ddot{y}_c - \ddot{\delta}$$

where δ is the deflection of the spring of a simple sprung mass model, ω_n (rad/sec) is the natural frequency of the model, and ζ is the damping ratio.

Model Number	ζ	ω_n (rads/sec)	Name
1	0.224	52.9	Spinal
2	0.40	25.1	Visceral
3	1.0	52.9	Body Vibration
4	1.0	1.571	Low Frequency

The VIBRATION RIDE QUALITY INDEX (VRQI) is defined as

$$\text{VRQI} = \frac{\ddot{y}_n' \text{ (RMS)}}{g}$$

where $\ddot{y}_n'$(RMS) is the maximum value obtained from one of the four model outputs.

The proposed limits on VRQI are as follows:

Limit	Description	VRQI must be less than:
A	Severe, less than one hour	0.5
B	Tolerable, less than one hour	0.2
C	Long-term, severe	0.2
D	Long-term, tolerable	0.1

The IMPACT RIDE QUALITY INDEX (IRQI) is obtained from the "DRI" output of Model Number 1, the "spinal model."

$$\text{DRI} = \frac{\omega_1^{\,2}\delta_{1\ MAX}}{g}$$

$\delta_{1\ MAX}$ is computed for each maximum value; i.e. each δ_1 when $\dot{\delta}_1 = 0$, $\ddot{\delta}_1 < 0$

We now order the DRI values as in the following example:

DRI	Number of Occurrences/hr.	DRI Exceedance Point	Number of Exceedances/hr.	Corresponding Number of Exceedances in 24 hrs.
0 -0.5	109	0	149	3576
0.5-1.0	9	0.5	40	960
1.0-1.5	10	1.0	31	744
1.5-2.0	7	1.5	21	504
2.0-2.5	7	2.0	14	336
2.5-3.0	4	2.5	7	168
3.0-3.5	2	3.0	3	72
3.5-4.0	1	3.5	1	24
4.0-4.5	0	4.0	0	0
4.5-5.0	0	4.5	0	0

The exceedances per twenty-four hours are obtained by ratioing up from the duration for which readings were actually obtained; in this example, one hour. The exceedance points are then plotted as shown ("Example A") in Figure 16. The IRQI is defined as the largest value which occurs. In Example A, IRQI = 1.0, so that the ride is just at the limit of tolerability. In Example B, the maximum value is about IRQI = 0.38, indicating a relatively smooth ride.

Appendix I. A Comparison With Experiment

It is hoped that, in the future, crew performance will be measured at sea, related to ship motions, and then compared with the proposed models. At the time of writing this paper, we have only had the opportunity to compare the model with one set of laboratory data, as shown in Figure I-1.

Appendix II. Response of a Linear, Damped Second Order System to Sinusoidal Vibration

It is the purpose of this Appendix to define the various responses of a linear damped dynamic model to sinusoidal vibration.

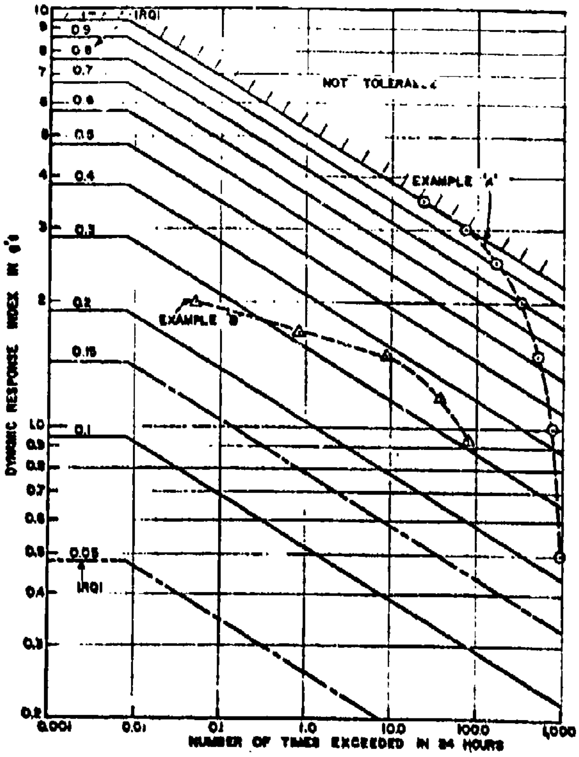


Figure 16. Impact Ride Quality Index Chart. (The two examples of exceedance plots are hypothetical.)

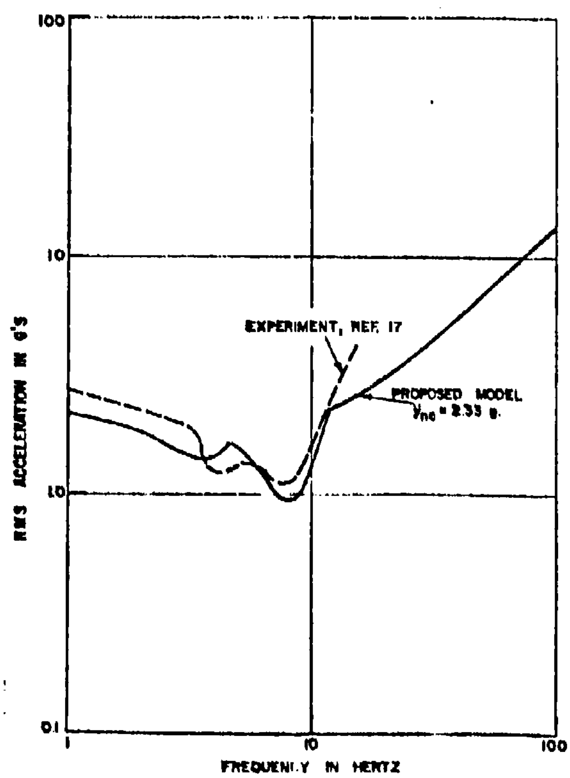
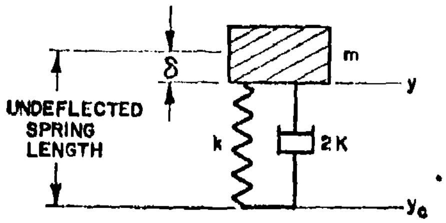


Figure I-1. "One minute" tolerance to sinusoidal vibration [Ziegenruecker & Magid(17)] compared with the proposed model, $\ddot{y}_{n_{crit}} = 2.33g$.



Let

k = spring stiffness, lb/ft of compression

$2K$ = viscous damping constant
  = damper force/velocity (lb sec/ft)

δ = spring (compressive) deflection (ft)
  $= \lambda - (y - y_c)$

λ = unloaded spring length (ft)

m = mass (slugs)

F_s = force in the spring (lb) $(=k\delta)$

F_D = force in the damper (lb) $(=2K\dot{\delta})$

The equation of motion for mass m is

$$F_D + F_s = m\ddot{y} = 2K\dot{\delta} + k\delta \qquad \text{(II-1)}$$

But

$$y = \lambda + y_c - \delta$$

$$\therefore \ddot{y} = \ddot{y}_c - \ddot{\delta}$$

$$m\ddot{\delta} + 2K\dot{\delta} + k\delta = m\ddot{y}_c \qquad \text{(II-2)}$$

Dividing throughout by m, and writing

$$c = \frac{K}{m}$$

$$\omega^2 = \frac{k}{m}$$

$$\ddot{\delta} + 2c\dot{\delta} + \omega^2\delta = \ddot{y}_c \qquad \text{(II-3)}$$

For the special case when the model is subjected to a sinusoidal motion

$$\ddot{y}_c = \ddot{Y}_o \sin \Omega t \qquad \text{(II-4)}$$

where $\ddot{Y}_o = \Omega^2 Y_o$, the input acceleration amplitude. We know that the nontransient solution will be:

$$\left.\begin{aligned} \delta &= \delta_m \sin(\Omega t - \theta) \\ &= \delta_m \sin \psi \text{ (say)} \\ \therefore \dot{\delta} &= \Omega\delta_m \cos \psi \\ \ddot{\delta} &= -\Omega^2\delta_m \sin \psi \end{aligned}\right\} \qquad \text{(II-5)}$$

where δ_m is the motion amplitude and the phase angle θ has not been determined, we also note that

$$\ddot{Y}_o \sin \Omega t = \ddot{Y}_o \sin(\psi + \theta)$$

$$= \ddot{Y}_o (\sin \psi \cos \theta + \cos \psi \sin \theta) \qquad \text{(II-6)}$$

Substituting equations (II-5) and II-6) in (II-4), and equating the sin ψ and cos ψ coefficients separately

$$\left.\begin{aligned} -\Omega^2\delta_m + \omega^2\delta_m &= \ddot{Y}_o \cos \theta \\ 2c\Omega\delta_m &= \ddot{Y}_o \sin \theta \end{aligned}\right\} \qquad \text{(II-7)}$$

This defines the phase angle, i.e.

$$\tan \theta = \frac{2c\Omega}{\omega^2 - \Omega^2} = \frac{2\bar{c}(\Omega/\omega)}{1 - (\Omega/\omega)^2} \qquad \text{(II-8)}$$

where $\bar{c} = c/\omega$, the critical damping ratio. Squaring and adding equation (II-7) gives

$$\delta_m^2[(2c\Omega)^2 + (\omega^2 - \Omega^2)^2] = \ddot{Y}_o^2$$

or

$$\frac{\delta_m}{\ddot{Y}_o} = \frac{1}{\sqrt{(2c\Omega)^2 + (\omega^2 - \Omega^2)^2}} \qquad \text{(II-9)}$$

alternatively

$$\frac{\omega^2\delta_m}{\ddot{Y}_o} = \frac{k\delta_m}{m\ddot{Y}_o} = \frac{1}{\sqrt{[(2\bar{c}(\Omega/\omega)]^2 + [1 - (\Omega/\omega)^2]^2}} \qquad \text{(II-10)}$$

$$= \frac{\text{peak spring force } F_{s_{max}}}{\text{mass} \times \text{peak input acceleration}}$$

The total force is

$$F_T = F_s + F_D = m(2c\dot{\delta} + \omega^2\delta) \qquad \text{(II-11)}$$

Substituting equation (II-5) for δ and $\dot{\delta}$

$$\frac{F_T}{m} = 2c\Omega\delta_m \cos \psi + \omega^2\delta_m \sin \psi$$

$$\therefore \frac{F_{T_{max}}}{m} = \delta_m \sqrt{(2c\Omega)^2 + \omega^4} \qquad \text{(II-12)}$$

$$= \ddot{Y}_o \sqrt{\frac{(2c\Omega)^2 + \omega^4}{(2c\Omega)^2 + (\omega^2 - \Omega^2)^2}} \qquad \text{(II-13)}$$

from equation (II-9).

$$\therefore \frac{F_{T_{max}}}{m\ddot{Y}_o} = \sqrt{\frac{1 + [2\bar{c}(\Omega/\omega)]^2}{[2\bar{c}(\Omega/\omega)]^2 + [1 - (\Omega/\omega)^2]^2}} \qquad \text{(II-14)}$$

$$= \frac{\text{peak total force } (F_D + F_s)_{max}}{\text{mass} \times \text{peak input acceleration}}$$

$$= \frac{\ddot{y}_{max}}{\ddot{Y}_o} = \frac{y_{max}}{Y_o} = \text{relative amplitude of } y \text{ to } y_c$$

(This last relationship follows from the fact that $m\ddot{y} = F_T$.)

The power dissipated by the damper is

$$P(t) = 2K(\dot{\delta})^2 = 2K\Omega^2\delta_m{}^2 \cos^2\psi \qquad \text{(II-15)}$$

Averaging this over one cycle

$$P = K\Omega^2\delta_m{}^2 = \frac{mc\Omega^2\ddot{Y}_o{}^2}{[(2c\Omega)^2 + (\omega^2 - \Omega^2)^2]} \qquad \text{(II-16)}$$

or

$$\frac{\omega P}{m\ddot{Y}_o{}^2} = \frac{\overline{c}\,(\Omega/\omega)^2}{[2\overline{c}(\Omega/\omega)^2] + [1 - (\Omega/\omega)^2]^2} \qquad \text{(II-17)}$$

Summary of Equations

Equations (II-10), (II-14) and (II-17) tell us all that is known about the model's behavior. Each of the parameters (maximum spring force or deflection, maximum total force or acceleration, or power absorbed) can be expressed as a constant [which is a function of $\overline{c}$ and (Ω/ω)] multiplied by the amplitude of the input acceleration $\ddot{Y}_o$. These constants are cited below and are illustrated in Figures II-1-3. They are of interest because ride conditions are often expressed in terms of "tolerable" values of $\ddot{Y}_o$ or the R.M.S. value of $\ddot{Y}$.

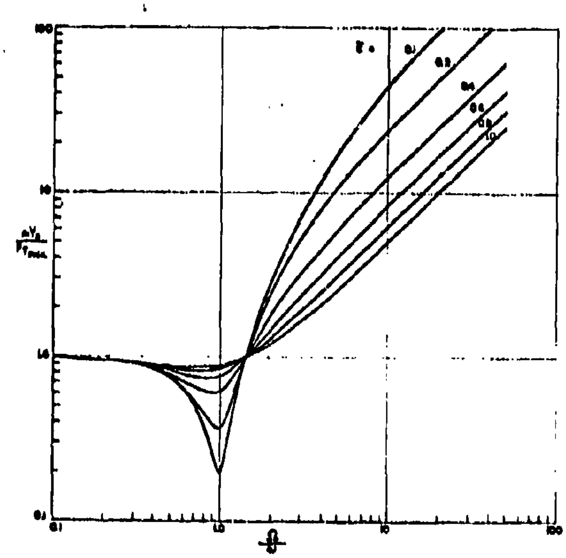


Figure II-1. Acceleration for constant total force (or mass acceleration) as given by equation (II-19).

Constant Maximum Spring Force (DRI Model)

$$\frac{\ddot{Y}_o}{\omega^2\delta_{max}} = \sqrt{[2\overline{c}(\Omega/\omega)]^2 + [1-(\Omega/\omega)^2]^2} \qquad \text{(II-18)}$$

Constant Maximum Total Force (Or Constant Maximum Acceleration)

$$\frac{m\ddot{Y}_o}{F_{T_{max}}} = \sqrt{\frac{[2\overline{c}(\Omega/\omega)]^2 + [1-(\Omega/\omega)^2]^2}{1 + [2\overline{c}(\Omega/\omega)]^2}} \qquad \text{(II-19)}$$

$$(= \ddot{Y}_o/\ddot{y}_{max})$$

This expression is plotted in Figure II-1 for a range of values of $\overline{c}$.

Constant Power Absorbed

$$\frac{\ddot{Y}_o}{\sqrt{\omega P/m}} = \frac{\sqrt{[2\overline{c}(\Omega/\omega)]^2 + [1-(\Omega/\omega)^2]^2}}{(\Omega/\omega)\sqrt{\overline{c}}} \qquad \text{(II-20)}$$

This expression is plotted in Figure II-2 and, for comparison examples of each of these three equations are plotted in Figure II-3 against Ω/ω for a value of $\overline{c}$ = .224 which is representative of the human torso. Their limits are shown in the table below.

	Constant Maximum Spring Force	Constant Maximum Total Force	Constant Power
Parameter	$\ddot{Y}_o/\omega^2\delta_m$	$m\ddot{Y}_o/F_{T_{max}}$	$\ddot{Y}_o/\sqrt{\omega P/m}$
Value at $\Omega/\omega = 0$	1	1	∞
$(\Omega/\omega)^2$ for minimum value	$(1-4\overline{c}^2)$	$\frac{\sqrt{1+8\overline{c}^2}-1}{4\overline{c}^2}$	-
Minimum Value of the Parameter	$2\overline{c}$	-	-
Parameter value at $\Omega/\omega = 1$	$2\overline{c}$	$\frac{2\overline{c}}{\sqrt{1+4\overline{c}^2}}$	$2\sqrt{\overline{c}}$
As $\Omega/\omega \to \infty$ Parameter $\to$	$(\Omega/\omega)^2$	$\frac{1}{2\overline{c}}(\Omega/\omega)$	$\frac{1}{\sqrt{\overline{c}}}(\Omega/\omega)$

It's of interest to note that the asymptotic power equation (as $\Omega/\omega \to \infty$)

$$\ddot{Y}_o = \sqrt{\omega P/m\overline{c}}\,(\Omega/\omega)$$

$$P = \frac{m\overline{c}}{\omega}\ddot{Y}_o{}^2(\omega/\Omega)^2 = mc\frac{\ddot{Y}_o{}^2}{\Omega^2} = \frac{K\ddot{Y}_o{}^2}{\Omega^2} \qquad \text{(II-21)}$$

Thus, power absorption for a given acceleration amplitude is less at the higher frequencies.

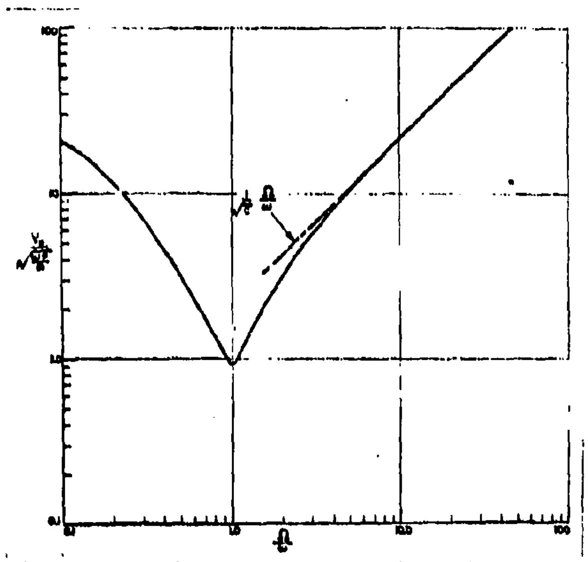


Figure II-2. The constant "power absorption" parameter of equation (II-20).

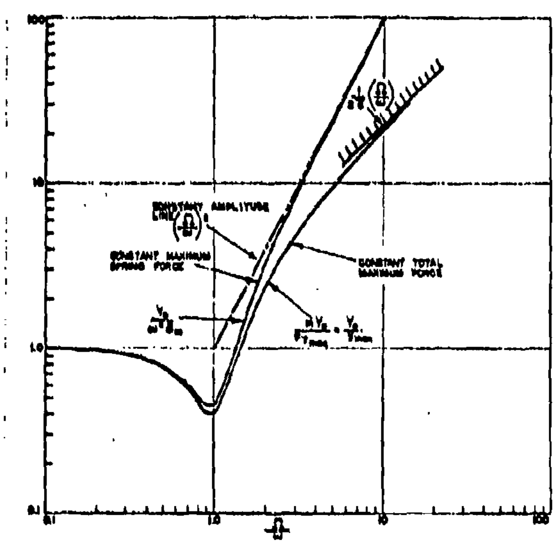


Figure II-3. The two vibration tolerance parameters for $\tau = 0.224$.

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Acknowledgements

Any scientific contribution rests on the earlier contributions of countless workers. In addition to this, the present study has been particularly helped by my long-time colleagues in biodynamics, Dr. Henning E. von Gierke,* Mr. James W. Brinkley,** Mr. Henry R. Jex of Systems Technology, Inc., Mr. Geoff Allan of the Royal Aircraft Establishment and his associate, Ms. Kate Maslen. This is not to imply that they agree with all I've written, of course, or that they endorse any or all of it.

The work would not have been done without the perceptive realization of the need for it in the Navy's Advanced Naval Vehicles Concept Evaluation Program by Captain Thomas L. Meeks† and Mr. Peter J. Mantle††. Thanks are also due to Mr. Robert A. Johnson of NAVSEC (Code 6136) for funding a study of the low-frequency end of the problem, below one Hertz and to Mr. Michael E. McCauley of Human Factors Research Inc. for helpful criticism and advice on kinetosis. Finally, I owe a debt to my colleagues at Payne, Inc., particularly Mr. E.G.U. Band and Mr. David R. Lavis for many helpful discussions.

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